7N-37
198735
P29

NASA TN D-276

TECHNICAL NOTE

D-276

PERFORMANCE STUDY OF A PISTON-TYPE PUMP FOR LIQUID HYDROGEN

By Arnold E. Biermann and William G. Shinko

Lewis Research Center
Cleveland, Ohio

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

WASHINGTON March 1960

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

TECHNICAL NOTE D-276

PERFORMANCE STUDY OF A PISTON-TYPE PUMP FOR LIQUID HYDROGEN

By Arnold E. Biermann and William G. Shinko

E-688

CG-1

SUMMARY

Data covering the performance of a low-speed, submerged, piston-type pump for handling boiling hydrogen are presented. Tests were made with JP-5 fuel, liquid nitrogen, and liquid hydrogen at flows up to 78 gallons per minute, speeds to 360 rpm, and with pressures to 130 pounds per square inch gage.

The overall performance of the pump was satisfactory with all fluids pumped. The effect on pump delivery of subcooling or of liquid level above the pump was slight. Internal leakage (slip) was greatest with hydrogen and least with JP-5 fuel. The cavitation losses were slightly greater with nitrogen than with hydrogen. The mechanical endurance of the pump in its final configuration was satisfactory for the intended application.

INTRODUCTION

As part of a research program involving liquefied gases, a pump was required for handling liquid hydrogen at pressures up to 130 pounds per square inch gage and flows to 55 gallons per minute. A further requirement stipulated that the pump be capable of handling the boiling fluid and, for reasons of safety and control, that it be driven by a hydraulic motor.

Several types of displacement pumps were considered for this application. The piston pump was selected because of its independence of pressure on speed, small leakage areas, low shaft speeds, and low intake velocities.

In the design of the pump, low specific bearing loads, as compared with standard practice, were employed in order to insure satisfactory submerged operation of the pump in liquid hydrogen. Submerged operation eliminates the need for a separate pump housing and avoids the accompanying heat leak to the fluid. In addition, submerged operation permits the vapor to separate from the liquid during the induction process.

Leakage areas are of particular concern in displacement pumps when pumping liquid hydrogen because of the low viscosity of the fluid. Leakage passing between parallel surfaces is inversely proportional to the viscosity. The viscosity of liquid hydrogen is approximately 27 percent less than that of air at room temperature and is roughly 100 times less than that of JP-5 fuel. Thus, the leakage of liquid hydrogen as caused by its low viscosity will be much greater than that of JP-5. Because of these high leakage potentials it is important that known leakage areas be reduced to a minimum. In the piston pump, the leakage areas are small as compared with the leakage areas found in many rotary pumps. In addition, the circular shape of the piston and cylinder provides some freedom from thermal distortion and permits a high degree of accuracy of fabrication.

E-688

A preliminary study of the subject piston pump is reported in reference 1. In this work, visual studies of cavitation at various piston speeds were made with a small model pump having a glass cylinder. The main objective of these tests was to determine by visual and photographic means the maximum piston speed as limited by cavitation. The tests also provided information regarding valve operation, inertia ram, and the effects of cavitation on the subcooling of the liquid.

These preliminary tests indicated satisfactory pumping under boiling conditions for both liquid nitrogen and liquid hydrogen. There was little observable difference in pump performance between liquid nitrogen and liquid hydrogen. For the limited conditions of the tests the mechanical functioning of the pump was satisfactory.

Reference 1 also discusses the phenomenon of subcooling by cavitation, which was obtained with liquid hydrogen and liquid nitrogen. A disk, representing a piston, was reciprocated in the boiling liquids sufficiently fast to cause cavitation. In this manner the liquid was cooled by evaporation through a reduction in pressure. Subcooling of approximately 0.6° F was obtained from the cavitation in boiling nitrogen and in boiling water. The subcooling obtained in a similar manner with hydrogen was less. In this connection it should be observed that subcooling of only 0.1° R is approximately equal to a head of 110 inches of liquid hydrogen or 5 inches of liquid nitrogen.

The purpose of the work reported herein is to provide more complete information concerning the performance of this pump over a wider range of conditions than that of reference 1. The work was done at the Lewis Research Center during 1959.

APPARATUS AND METHODS

Piston-Type Pump

The piston pump, shown in figures 1 to 5, is of the bucket type, in which the inlet valves are located in the pistons and the discharge valves are located in the cylinder heads. The five pistons of this pump are driven by a wobbler, Z-crank mechanism in which the torque reaction of the wobbler is taken by a ball-bearing roller that oscillates in a hardened steel guide. The cylinders are mounted on a plate that separates the common discharge chamber from the inlet side of the pump.

Plate valves of identical construction were employed for both the inlet and discharge. Valve port area for each valve was 35 percent of the piston area.

The design specifications of this pump are as follows:

Bore and stroke, in.	3 by 1.5
Total displacement, cu in.	53.03
Design speed, rpm	240
Displacement at design speed, gal/min	55.1
Design inlet conditions	Saturated liquid with zero head
Discharge pressure, lb/sq in. gage	130
Average piston speed, ft/sec	1.0
Lubrication (entire pump submerged in pumped fluid)	None
Total piston clearance, diametrical, in.	0.002
Operating temperature, $^{\circ}F$	-430

A detailed description of this pump presented in reference 1 includes a plot showing the variation of piston movement and the net variation of displacement with respect to angle of rotation.

No attempt was made to balance the pump mechanism because the design speed was low enough to avoid unbalanced forces of noticeable magnitude.

The ball bearings used throughout the pump were of commercial grade (SAE 52100 steel, nonstainless). The bearing surfaces were sprayed with a thin coat of light penetrating oil to reduce corrosion.

The parts of the pump subject to wear were hardened to Rockwell C of approximately 58. The cylinders, pistons, cylinder heads, pins, piston ball joints, and crankshaft were fabricated of stainless steel (AISI-304) with the wearing surfaces commercially nitrided. The wobbler, valve plates, and frame were also fabricated of AISI-304 steel; the discharge cover was made of AISI-347. The torque reaction guide was fabricated of AISI-440-C hardened to Rockwell C-58.

Several modifications of the spherical bearings used at the wobbler end of the piston rod were made during the testing program and are shown in order of use in the following table:

SPHERICAL BEARING CONFIGURATIONS USED

Ball	Outer race	Remarks
(A) AISI-440 steel, chrome plated	AISI 303 steel	Used during all preliminary tests. These tests were made at reduced loads and speeds. Failed by seizure in liquid nitrogen.
(B) SAE 52100 steel, chrome plated	52100 steel	Approx. 60% more bearing area than for (A). Failed by seizure in liquid nitrogen.
(C) SAE 52100 steel, chrome plated	Bronze	Same bearing area as for (B). Used with and without: (1) Molybdenum disulfide lubricant (2) Light penetrating oil Failed by seizure in liquid hydrogen.
(D) SAE 52100 steel, chrome plated	Heat-treated stainless steel with woven Teflon fiber-glass bearing insert bonded to outer race	Approx. 20% less bearing area than for (A). No measurable wear after extensive operation in both liquid hydrogen and liquid nitrogen.

Piston rings were fabricated of Teflon compounded with a glass fiber filler and were forced against the cylinder wall with stainless-steel inner rings.

Thermal distortion and differential expansion problems were minimized by use of materials having similar thermal expansion characteristics. Welding and brazing were avoided in all parts involving critical clearances.

Pump Test Rig

The apparatus used for investigating the pump is shown in figures 6 to 8. The test pump was mounted at the lower end of a vertical-shaft housing, which was flange-mounted at the top of a 280-gallon Dewar tank. The tank was buried in the ground, outdoors, as shown in figure 6. The indicating instruments and the hydraulic power system were housed in an adjacent building. The Dewar was provided with vent and pressurization control means for furnishing tank pressures up to 55 pounds per square inch gage.

The pump was driven through a vertical shaft by a fixed-displacement (2.35 cu in.) piston-type hydraulic motor mounted on the top of the tank. The drive shaft was mounted on commercial ball bearings having large radial clearances. The shaft was sealed at the top of the tank with a double seal in which the space between seals was pressurized with helium to a value slightly greater than tank pressure.

The hydraulic motor and shaft assembly were calibrated for torque with a prony brake in conjunction with differential pressure measurements across the hydraulic motor. The torque measurements obtained from the calibration thus reflected only the torque applied to the pump and did not include the bearing and seal losses of the drive shaft.

A variable-displacement pump driven by an electric motor supplied hydraulic fluid to the hydraulic drive motor.

The pump was submerged in the liquid with the fluid entering the open end of the pump cylinders directly from the tank. From the pump the fluid passed first through a Venturi-type flowmeter and then through a turbine-type flowmeter and finally through a remote-operated, pump-discharge-pressure control valve. From the control valve the fluid was discharged through a baffle and screen arrangement that diffused the fluid laterally back into the free liquid of the tank of low velocities.

The flow indications from the turbine-type meter were generally in agreement with those from the Venturi. Because of its more consistent flow indications, the turbine meter was used throughout these tests.

The initial tests were carried out with JP-5 fuel in a tank (not shown) 5 feet in diameter filled to about 4 feet above the pump inlet.

Pump performance data were obtained at speeds and discharge pressures ranging from 60 to 360 rpm and from 30 to 130 pounds per square inch, respectively. Pump speed was controlled by varying hydraulic-supply-pump displacement. The discharge pressure of the test pump was controlled by varying the opening position of the pump discharge valve. Liquid temperature at the pump inlet was approximately 70° F during all tests with the JP-5 fuel. After these tests, the entire pumping system was installed in the low-temperature facility.

Prior to loading the Dewar tank with cryogenic fluids, the Dewar and test-pump fluid systems were vacuum-purged and then pressure-purged with helium gas. Thereafter, a slight positive pressure was maintained in the Dewar until tests were completed in order to prevent the outside air from contaminating the test fluids.

Pump tests with the cryogenic liquids were performed for two liquid states, which will be referred to as "subcooled" and "boiling."

The subcooled state is defined as the condition in the Dewar tank corresponding to a pressure of about 15 pounds per square inch above actual saturation pressure. This state was obtained by first establishing saturation conditions with the Dewar vented. The vent was then closed and the tank was pressurized with helium to approximately 15 pounds per square inch above vented Dewar pressure. The flow rate was recorded immediately after this pressure was reached. Throughout this procedure the pump was operated at test speed and discharge pressure.

The boiling state is defined as that condition of the cryogenic liquid in the Dewar tank after temperature equilibrium is reached with the Dewar vented to atmosphere. Temperature equilibrium was obtained with the pump operating at test conditions. Because of the circulation of the liquid in the Dewar by the pump, little temperature stratification of the liquid was evident. The liquid level was allowed to vary over a range of from 3 to 6 feet above the pump inlet for both the "subcooled" and the "boiling" test conditions. In certain tests the level was reduced to a few inches above the pump to study the effect of head.

RESULTS AND DISCUSSION

In reference 1, preliminary tests of the pump were first made with JP-5 fuel, liquid nitrogen, and liquid hydrogen at lower-than-design speeds and pressures. The pump performance appeared satisfactory, and the initial studies indicated that variation of liquid level above the pump had no appreciable effect on performance. The effect of inlet conditions on the performance of the pump is reexamined in this report with boiling liquid hydrogen and liquid nitrogen.

E-688

Effect of Inlet Conditions

Effect of liquid level. - With both boiling liquid hydrogen and liquid nitrogen, the liquid level was repeatedly lowered to pump inlet level. During this operation, pressure and delivery remained practically unaffected. In a typical case with liquid hydrogen, the pump delivery depreciated less than 3 percent as liquid level was reduced from 59 to 4 inches above the pump. During this change in level, delivery remained practically constant until there was just sufficient liquid to cover the inlet ports.

Effect of subcooling. - Figure 9 presents the delivery characteristics of the pump with boiling and subcooled fluids. These data were obtained by first operating the pump in the boiling fluid at atmospheric pressure, and then with the tank pressure increased 15 pounds per square inch by pressurizing with helium gas. These tests were conducted with an average liquid height of 60 inches above the pump. At a speed of 240 rpm, the increase in delivery of changing from boiling to nonboiling hydrogen is approximately 4 percent.

Because of the minor effects of liquid height and subcooling on performance and because the subject pump was designed for submerged operation in a boiling liquid, complete pump performance characteristics were only obtained with the cryogenic fluids under boiling conditions. This information is presented in the remainder of this report. Liquid level varied somewhat during these tests; however, this variation is believed to have had little effect on performance.

Delivery and Volumetric Efficiency

The delivery of a piston pump is generally less than the displacement of the pump because of: (1) failure of the valves to open and close at the proper instant, (2) internal leakage or slip through valves or around the piston, and (3) cavitation that causes the pump to fill with vapor instead of liquid.

Valve operation. - In the bucket-type pump, as reported herein, the inlet valves in the piston are actuated by inertia, spring forces, and fluid pressures. In order to avoid excessive valve pressure drop, the spring forces are made sufficient just to close the valves under static conditions. At the beginning of the inlet stroke, the inertia of the valve plate serves to open the valve port as the piston is accelerated. After midstroke, the piston decelerates and the valve plate tends to continue at midstroke velocity. Under these conditions alone, the valve would close before the piston reached the end of the stroke, and a loss in capacity would result. In this respect, the design philosophy is to rely on the fluid pressure to hold the valve open during the latter part of the stroke. Photographs in reference 1 show that the valve opening period was satisfactory at all piston speeds employed.

In the preliminary tests of the subject pump in reference 1, visual observation of the valves indicated that the discharge valves were closing late because of their inertia and lack of sufficient spring force. This late closure allowed back-flow into the cylinder during the intake stroke. The original springs were replaced with stronger ones for all subsequent tests as reported herein. As shown in figure 10, the use of the stronger springs increased the delivery of JP-5 fuel about 4 percent.

Piston and valve leakage. - Internal leakage, or slip, in small quantities is in itself not an important factor. If the leakage liquid, however, is converted to a gas after passing through the leakage restriction and if the gas is inducted back into the inlet of the pump, leakage becomes very important because of its effect on the volumetric efficiency of the pump. Heat is added to the liquid during the pumping process and, when the leakage fluid escapes into the inlet, a portion can change to vapor.

In the submerged bucket pump of this report, it is likely that some of the leakage vapor escaped directly into the tank without being inducted and that some of the vapor was condensed by inertia ram pressure.

Knowledge as to whether pump delivery suffers from slip or from cavitation losses can be obtained by operating the pump with liquids having different viscosity and vapor pressures. JP-5 fuel, because of its high viscosity (approx. 100 times that of hydrogen) is subject to very low leakage losses. In closed-discharge tests the measured leakage of JP-5 through the pump was less than one-tenth of 1 percent of rated flow. The low vapor pressure of JP-5 also results in low cavitation losses. Consequently, the delivery and volumetric efficiencies with JP-5, as shown by figures 10 and 11, are higher than with liquid nitrogen or liquid hydrogen.

Liquid nitrogen has about 10 times the viscosity of liquid hydrogen. Accordingly, the slip losses with liquid nitrogen, as shown by the left side of the curves in figure 11, are much less than for liquid hydrogen but more than for JP-5. The low volumetric efficiency obtained with hydrogen at low speeds is apparently caused by slip, and slip losses are accentuated by increase of discharge pressure.

Factors affecting cavitation characteristics. - In the conventional piston pump, the fluid is drawn into the cylinder during the suction stroke and is forced back out on the return stroke. During the suction stroke, the piston must accelerate the fluid in the intake system and then draw this fluid through the inlet valve. Both the acceleration of the fluid and its passage through the inlet valve cause a pressure drop that is reflected in the suction performance of the pump.

In the bucket-type pump, as reported herein, the fluid moves through the cylinder in one direction only. During the discharge stroke, fluid is accelerated and drawn into the cylinder on the inlet side of the piston. During the inlet stroke, the piston moves through the fluid in the cylinder. Any velocity change in the fluid is caused by the difference between flow area of the cylinder and that of the inlet valve. The pressure in the cylinder during the inlet stroke is thus determined by the pressure drop across the inlet valve alone; whereas, in the conventional piston pump, the pressure in the cylinder during the inlet stroke is a function of the pressure drop required to accelerate the fluid column in addition to the pressure drop across the valve. Thus, for comparable valve areas the pressure reduction in the cylinder of the bucket-type pump will be less, and the tendency to cavitate will be reduced as compared with that of the conventional pump.

During the discharge stroke of a bucket-type pump, the piston movement causes fluid to move toward the inlet side of the piston. This general flow toward pump inlet tends to persist after the discharge stroke and into the suction stroke. Whether the wave motion toward the piston persists long enough to effectively charge the cylinder is, of course, a function of the frequency of the wave motion with respect to the frequency of the pumping cycle. It is apparent that inertia ram can also be detrimental to charging if it is out of phase with the filling cycle.

Some insight into the effects of cavitation on pump delivery can be obtained by pumping different liquids at or near their boiling points. Nitrogen, because of its high density and relatively high viscosity, fails to follow the quick reversals of a piston, and vapor pockets are readily formed in the saturated fluid. Consequently, the volumetric efficiency with nitrogen decreases at the higher speeds, as shown in figure 11. Hydrogen, because of its low density and low viscosity, follows the piston movement readily, and vapor pockets are not as readily formed as with nitrogen. Consequently, with hydrogen the effect of pump speed on volumetric efficiency is not so marked. Furthermore, the lower specific heat of nitrogen causes boiling with less heat addition than for hydrogen.

Torque and Torque Efficiency

Pump torque and torque efficiency were obtained for the three test fluids at various pump speeds and pressures, as shown in figures 12 and 13. The torque required with JP-5 fuel was higher than with hydrogen or nitrogen in spite of the better lubricating qualities of JP-5. The effects of volume of vapor present in the cylinder on the torque of a piston pump are apparently so great as to completely overshadow small differences in friction. When handling cryogenic fluids, the pump cylinder is filled with a saturated mixture of gas and liquid during the intake stroke. Initial compression of the gas on the discharge stroke converts all gas to liquid with a very low pressure rise and little expenditure of work. At this point delivery starts and the work performed during the discharge stroke is mainly a function of the quantity of liquid trapped in the cylinder. Thus, in these tests torque was greatly affected by delivery, and the small differences in pump friction with the three test fluids were not easily detected.

In figure 12 the tendency of the curves for JP-5 and liquid nitrogen to rise with speed is apparently caused by the increase in drag of the moving parts of the pump from the higher density and viscosity of these fluids as compared with hydrogen.

Overall Efficiency

The dependency of torque on delivery makes torque efficiency an inadequate criterion of pump performance. The overall efficiency, however, obtained by multiplying volumetric efficiency and torque efficiency provides a good measure of torque for a given delivery (ref. 2). Figure 14 shows the overall efficiency for the three fluids tested. At 240 rpm and 130 pounds per square inch, the overall efficiency is approximately the same for all three fluids. At this rated pressure and at the higher speeds, the efficiency with both liquid hydrogen and liquid nitrogen is generally lower than for JP-5 fuel. At low speeds, the high percentage leakage with hydrogen apparently caused the drop in efficiency shown.

Pressure Developed at Vapor-Lock Conditions

The maximum pressure developed by a vapor-locked piston pump is the pressure developed by the pump acting as a gas compressor. The maximum delivery pressure of a piston-type compressor is obtained when the expanded volume of gas from the clearance volume equals the displacement volume. As long as the pressure developed by a liquid pump while pumping a gas is higher than the required delivery pressure, the pump will not vapor-lock completely.

In these tests the maximum pressure rise across the pump when pumping hydrogen gas was approximately 40 pounds per square inch. Higher pressures, as limited by vapor-lock can, of course, be obtained by decreasing piston clearance volume; however, in order to achieve simplicity of construction, little effort was made to obtain a minimum clearance volume. There is no indication that complete breakdown of pumping was imminent with any of the fluids tested. While pumping liquid hydrogen, the pump was stopped repeatedly by closing the outlet shutoff valve. Because of heat leak into the system, this procedure caused the cryogenic fluids to vaporize and fill the pump with gas. No difficulty or delay was experienced in resuming normal pumping operation after such stops.

Mechanical Durability

Pump wear. - After 9.7 hours of operation with JP-5 fuel, 6.5 hours with nitrogen, and approximately 22.8 hours with hydrogen, the wear of ball bearings, pistons, cylinders, and piston rings was very small. Tool marks were yet visible on piston ring surfaces.

Mechanical failures. - The ball joints that transmit the connecting-rod forces to the wobbler were a primary cause of failure. Without lubrication these joints were inadequate in any metal-to-metal combination tried for the range of loads covered during these tests. At the more severe load conditions, these joints galled and sometimes seized. In several instances the seizure of a ball joint caused failure of the connecting rod. The metal-to-metal ball joints were finally replaced with smaller joints fitted with a woven Teflon - fiber-glass lining (see table on p. 4). These joints were highly successful and showed no signs of wear after 6.4 hours of operation, of which the greater part was with liquid hydrogen.

In several instances the metal particles, which were a product of the abraded ball joints, were inducted into the cylinders and caused local penetration of the nitrided surfaces on the piston and cylinder. This permitted the two soft stainless undersurfaces to rub together, which caused local abrasion and eventual seizure.

One of the nitrided cylinders was replaced with a cast-iron cylinder and was operated in liquid hydrogen for a short period. At the finish of this test, the cylinder was badly abraded, and the tank and pump surfaces were covered with a grey iron powder that exhibited magnetized iron filing patterns on all steel surfaces capable of being magnetized.

Experience now indicates that a less vulnerable piston and cylinder combination would have been a hardened cylinder operating in conjunction with a piston fitted with side thrust surfaces of Teflon. This combination would have permitted close clearances without the possibility of galling from foreign particles.

SUMMARY OF RESULTS

From performance tests of a submerged, piston-type cryogenic pump the following concluding remarks are submitted:

1. The overall performance with JP-5 fuel, liquid nitrogen, and liquid hydrogen was satisfactory.

2. At 240 rpm the delivery with boiling hydrogen was 96 percent of that with subcooled hydrogen. Variation of liquid level height above the pump had little effect on pump delivery.

3. With the boiling fluid the volumetric efficiency at design conditions was 83, 86, and 96 percent with hydrogen, nitrogen, and JP-5 fuel, respectively. Internal leakage (slip) was greatest with hydrogen and least with JP-5, and explains the low volumetric efficiency with hydrogen.

4. Cavitation losses were slightly greater with nitrogen than with hydrogen.

5. At design conditions, the overall efficiency was approximately 83 percent for the three fluids tested.

6. Mechanical durability of the pump was satisfactory except for scuffing of pistons and cylinders from foreign materials and failure of the ball joints that connect the wobbler and the piston rods. The latter problem was resolved through the use of Teflon-lined ball joints.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, January 6, 1960

REFERENCES

1. Biermann, Arnold E., and Kohl, Robert C.: Preliminary Study of a Piston Pump for Cryogenic Fluids. NASA MEMO 3-6-59E, 1959.

2. Wilson, W. E.: Performance Criteria for Positive-Displacement Pumps and Fluid Motors. Paper No. 48-SA-14, ASME, 1948.

E-688

E-688

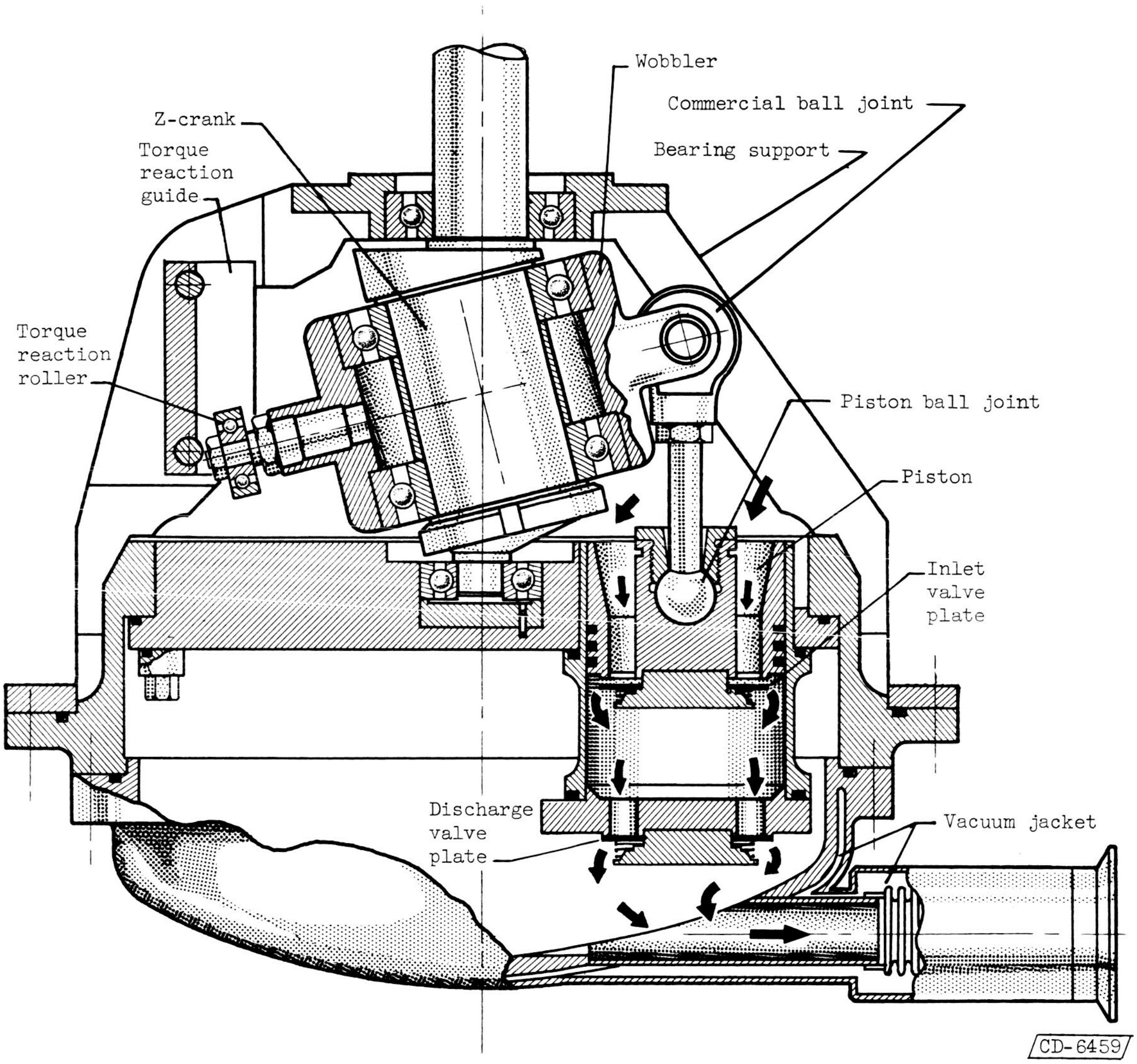


Figure 1. - Section through piston pump.

E-688

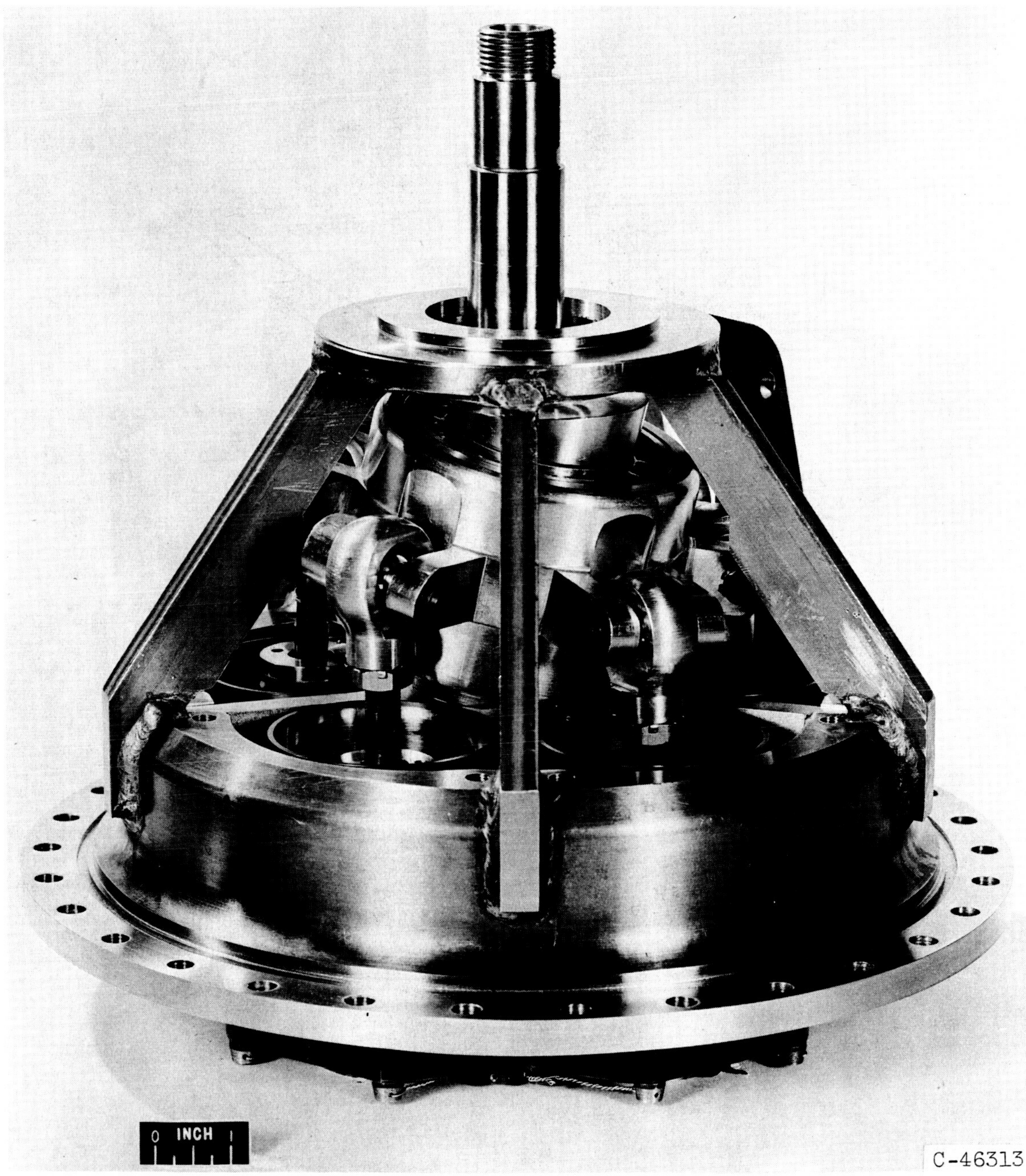


Figure 2. - Pump assembly with discharge cover removed as constructed for submerged operation in bottom of tank.

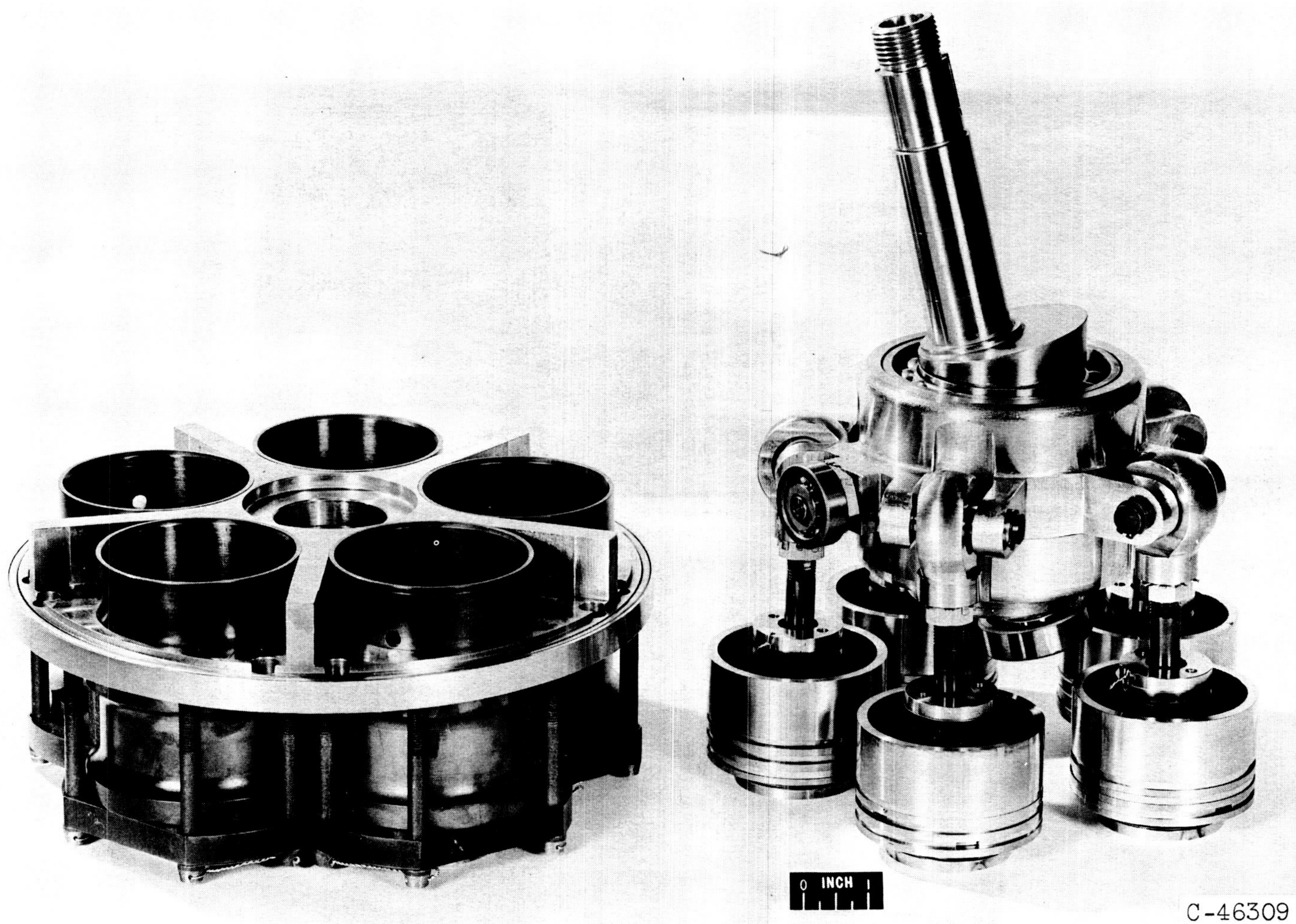


Figure 3. - Cylinder and piston assemblies.

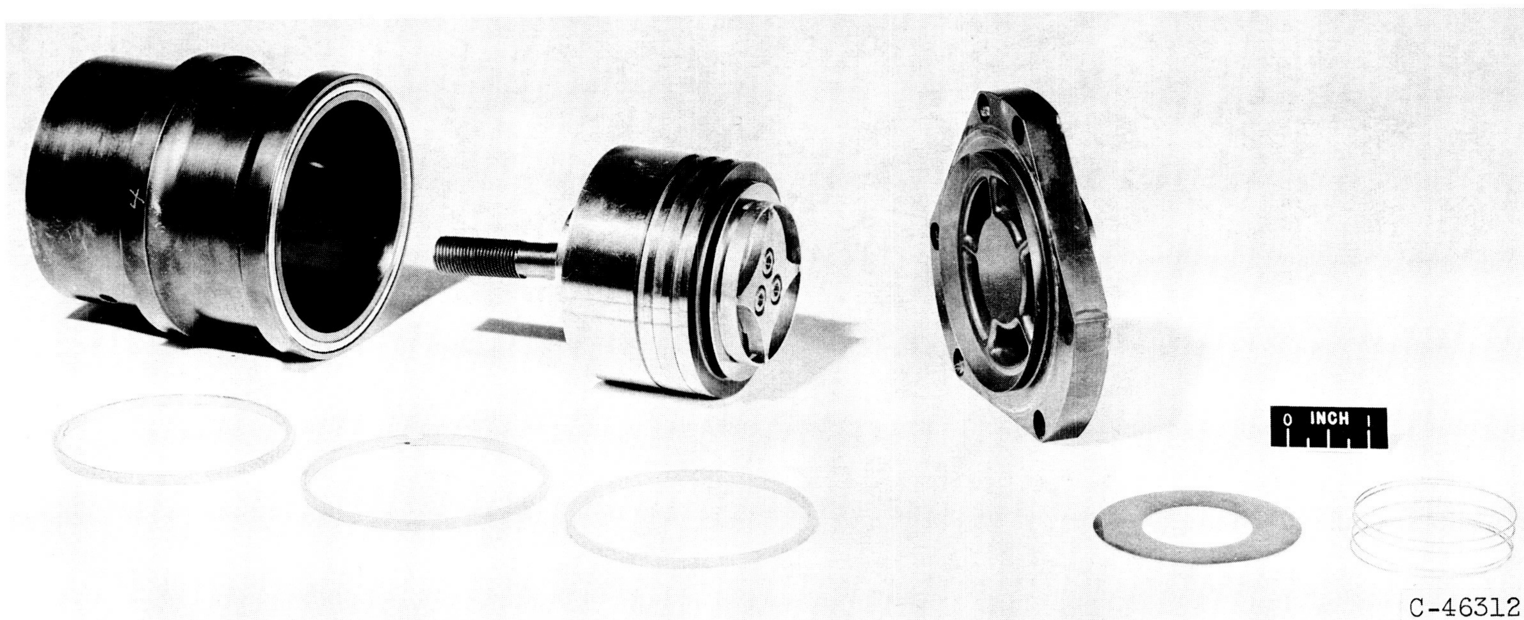


Figure 4. - Cylinder and piston parts.

E-688

CG-3

C-46311

Figure 5. - View showing discharge valves at bottom of cylinder assembly.

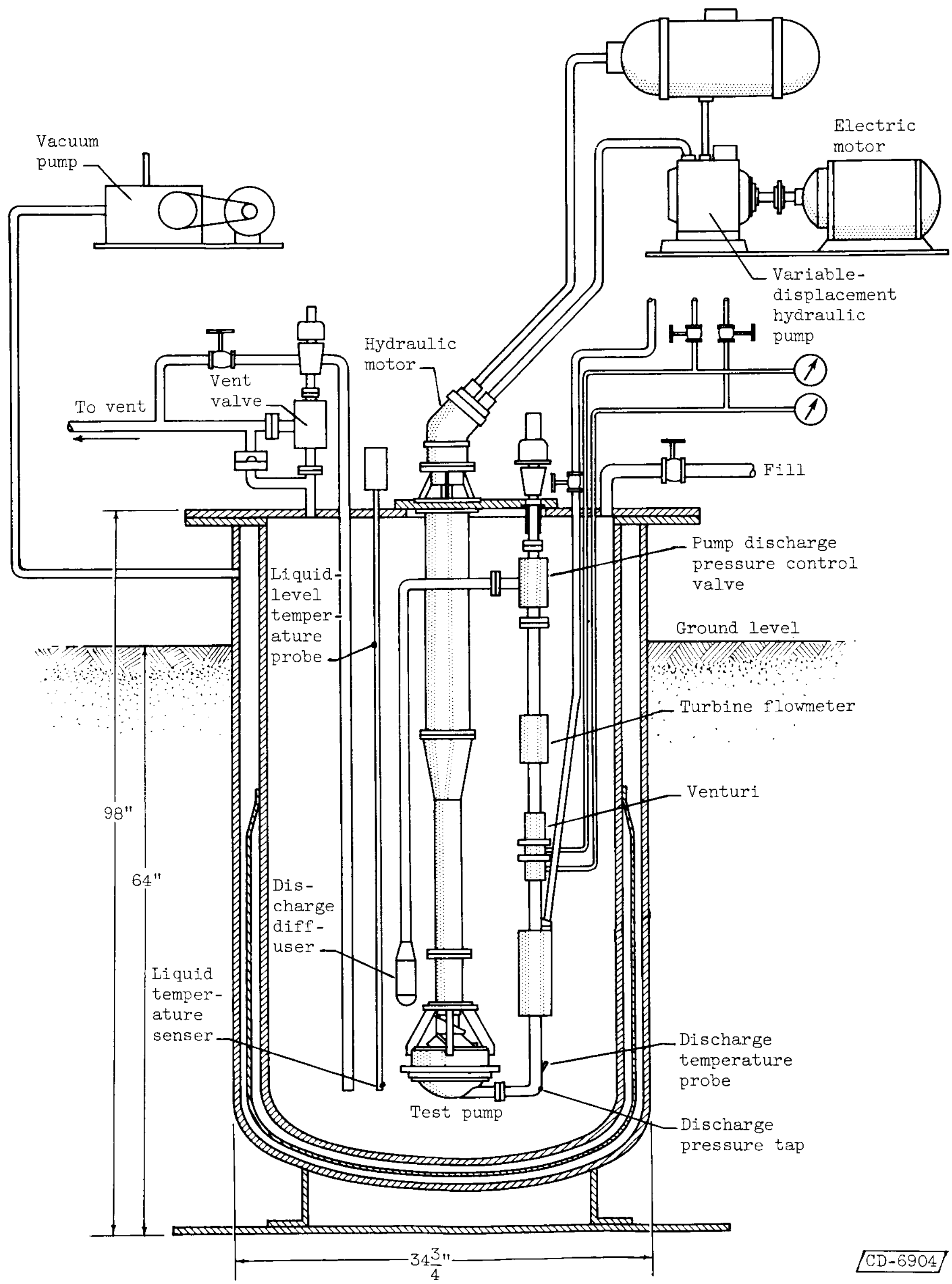


Figure 6. - Test rig.

E-688

E-688

CG-3 back

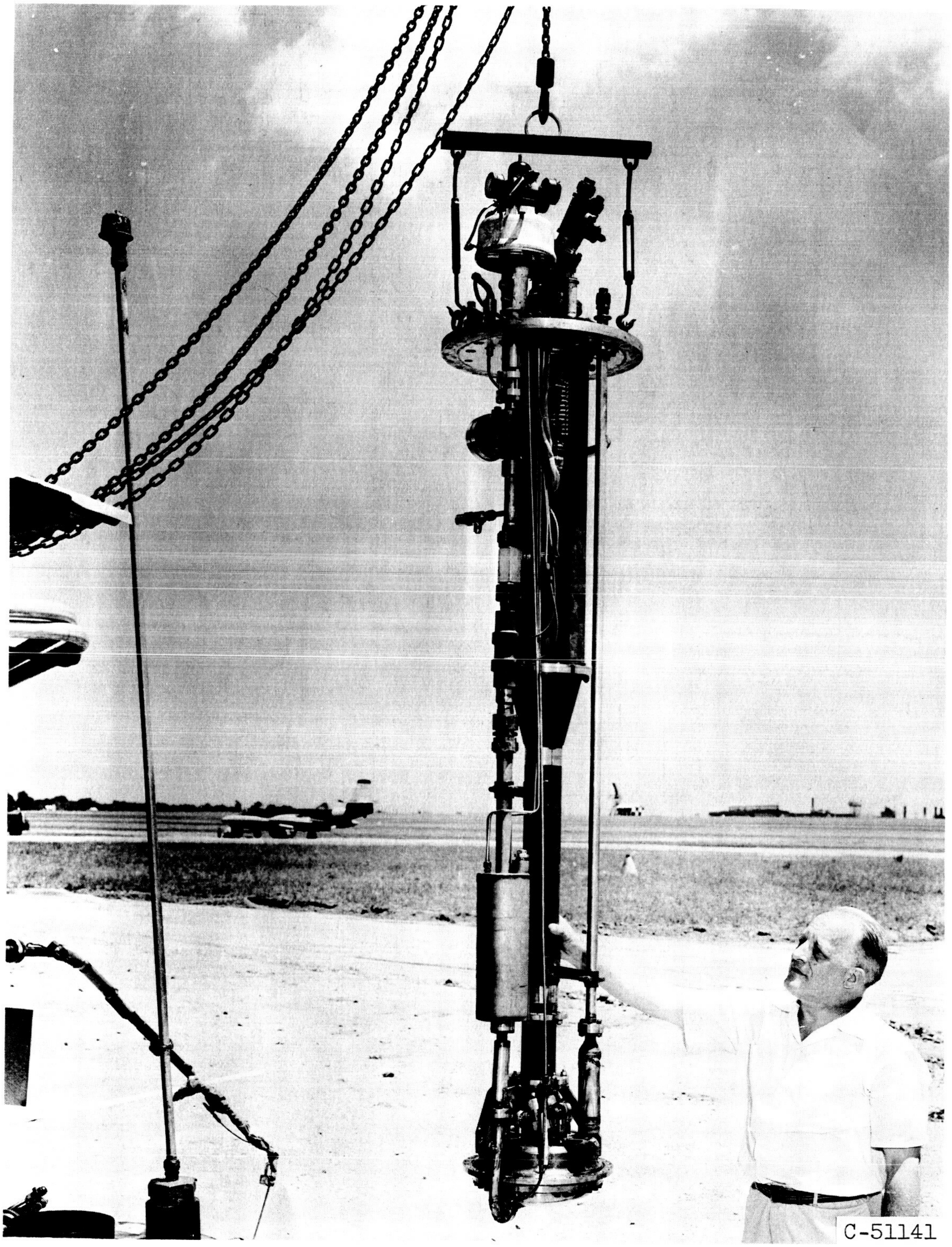


Figure 7. - Pump test assembly.

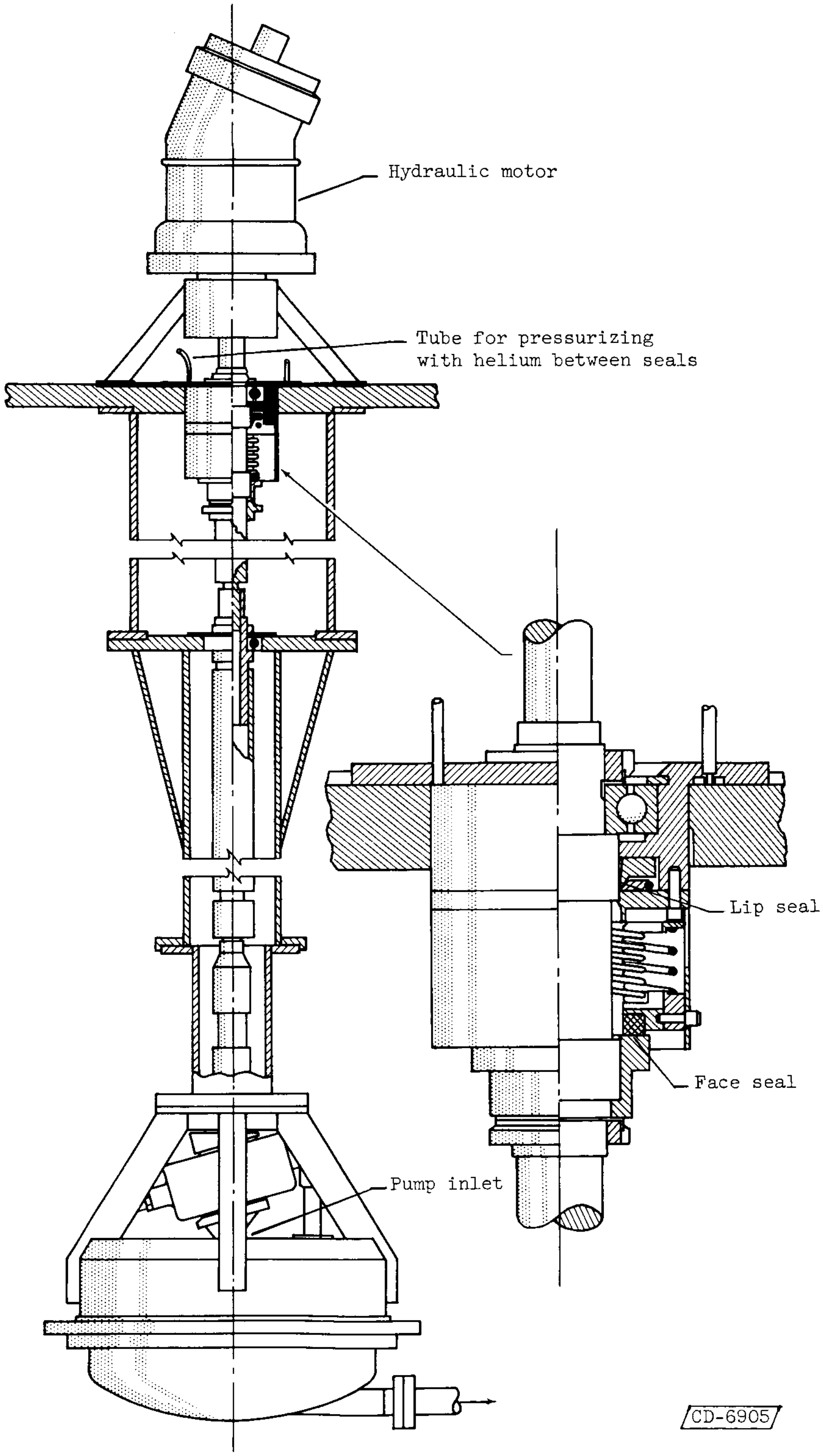
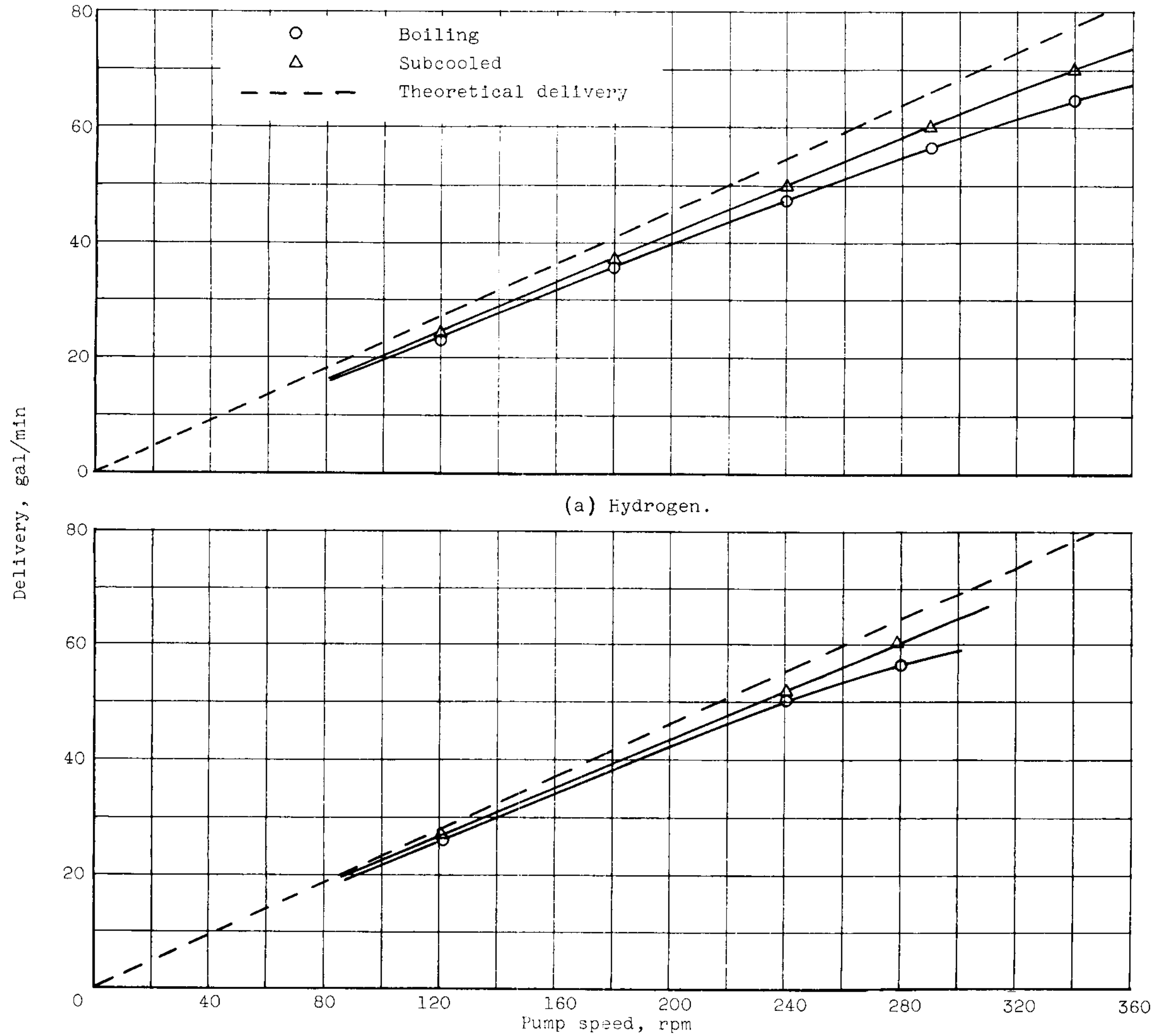


Figure 8. - Partial section showing pump drive-shaft housing and seal.



(a) Hydrogen.

(b) Nitrogen.

Figure 9. - Delivery with boiling and subcooled liquids and a discharge pressure, ΔP, of 50 pounds per square inch. Liquid level, approximately 60 inches above pump inlet.

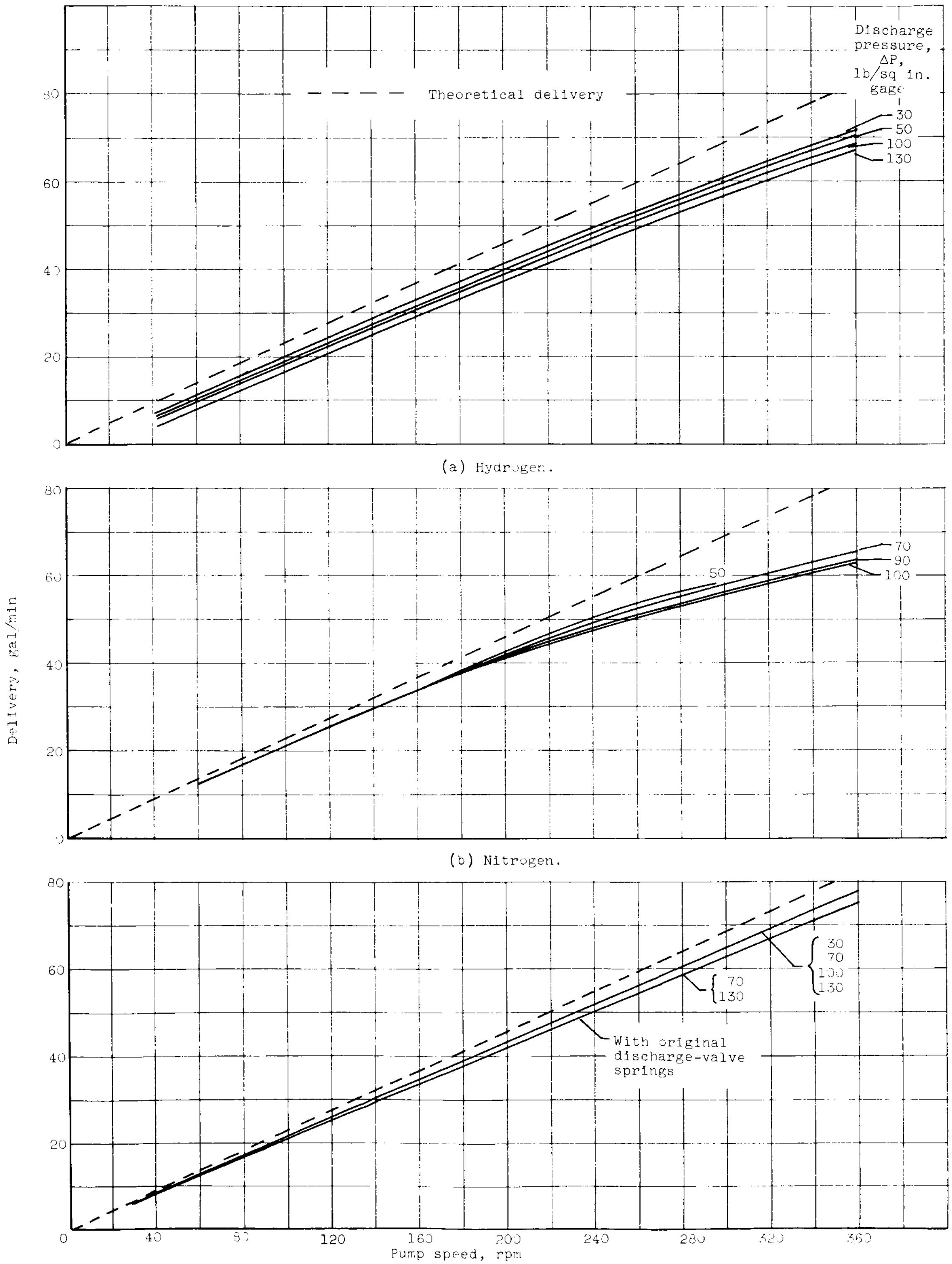


Figure 10. - Pump delivery with JP-5 fuel, boiling hydrogen, and boiling nitrogen. (Lower curve for JP-5 is with original springs for discharge valves. Remainder of program was run with stronger springs.)

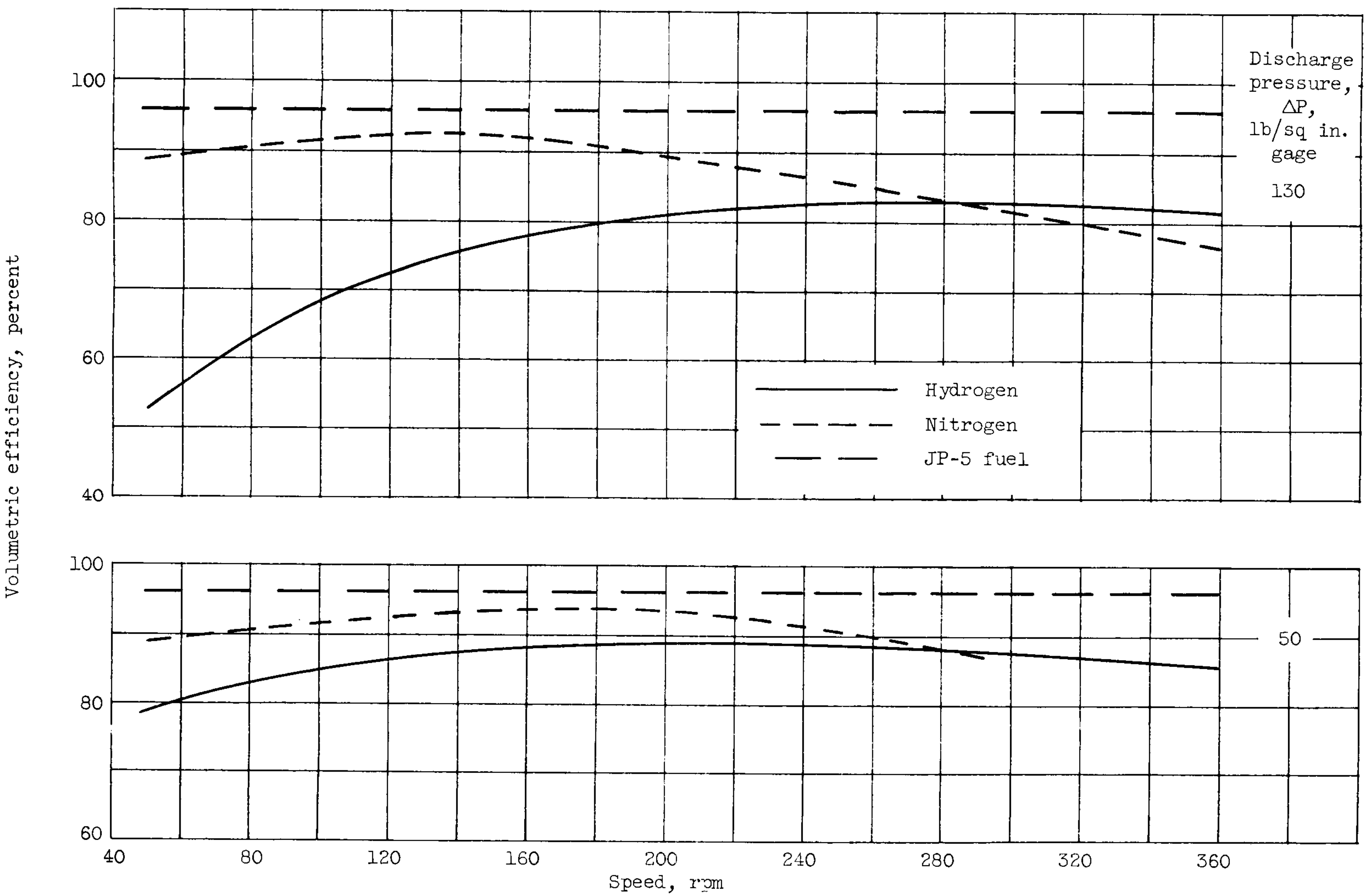


Figure 11. - Pump volumetric efficiency with JP-5 fuel, boiling hydrogen, and boiling nitrogen.

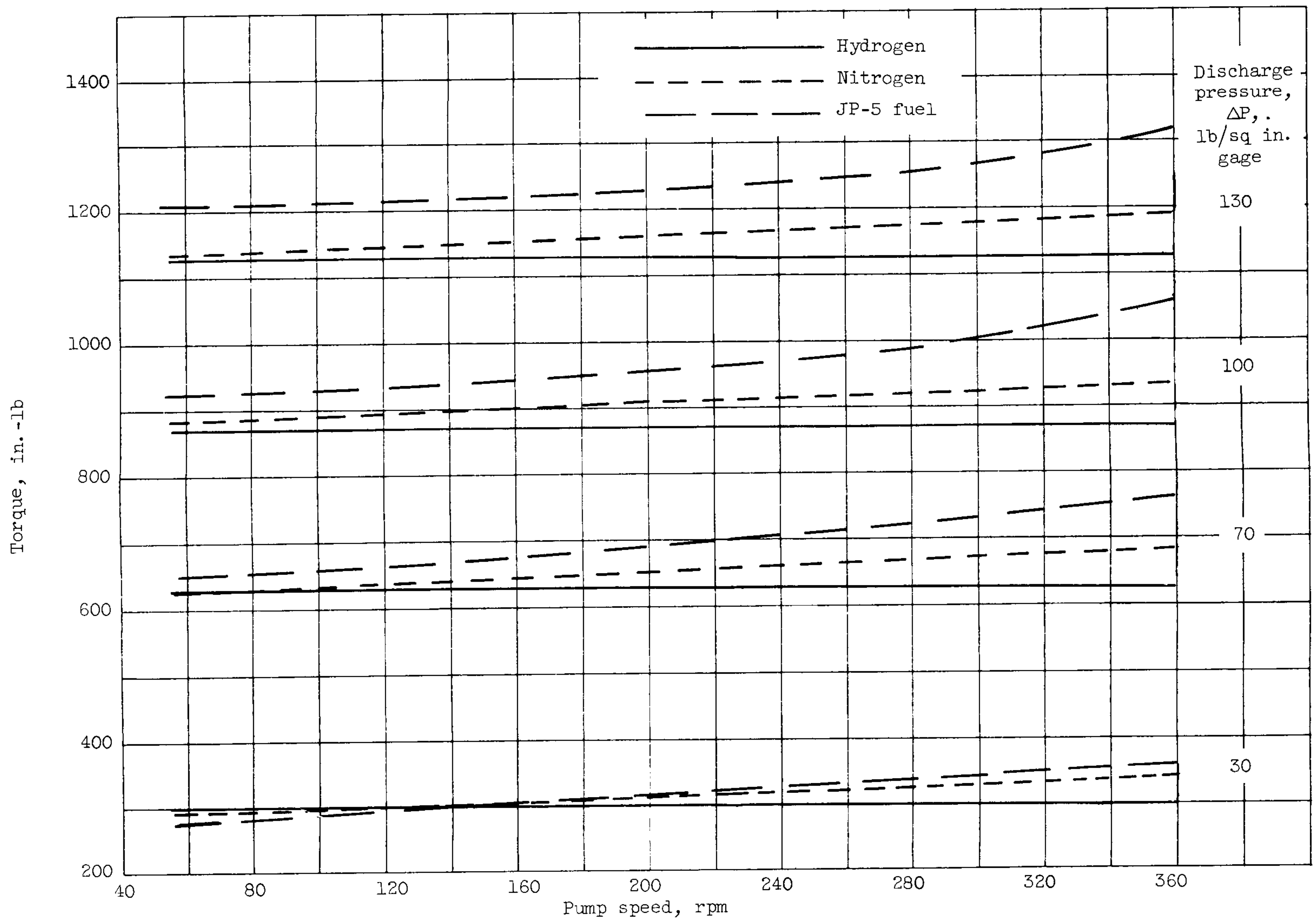


Figure 12. - Torque requirements with JP-5 fuel, boiling hydrogen, and boiling nitrogen.

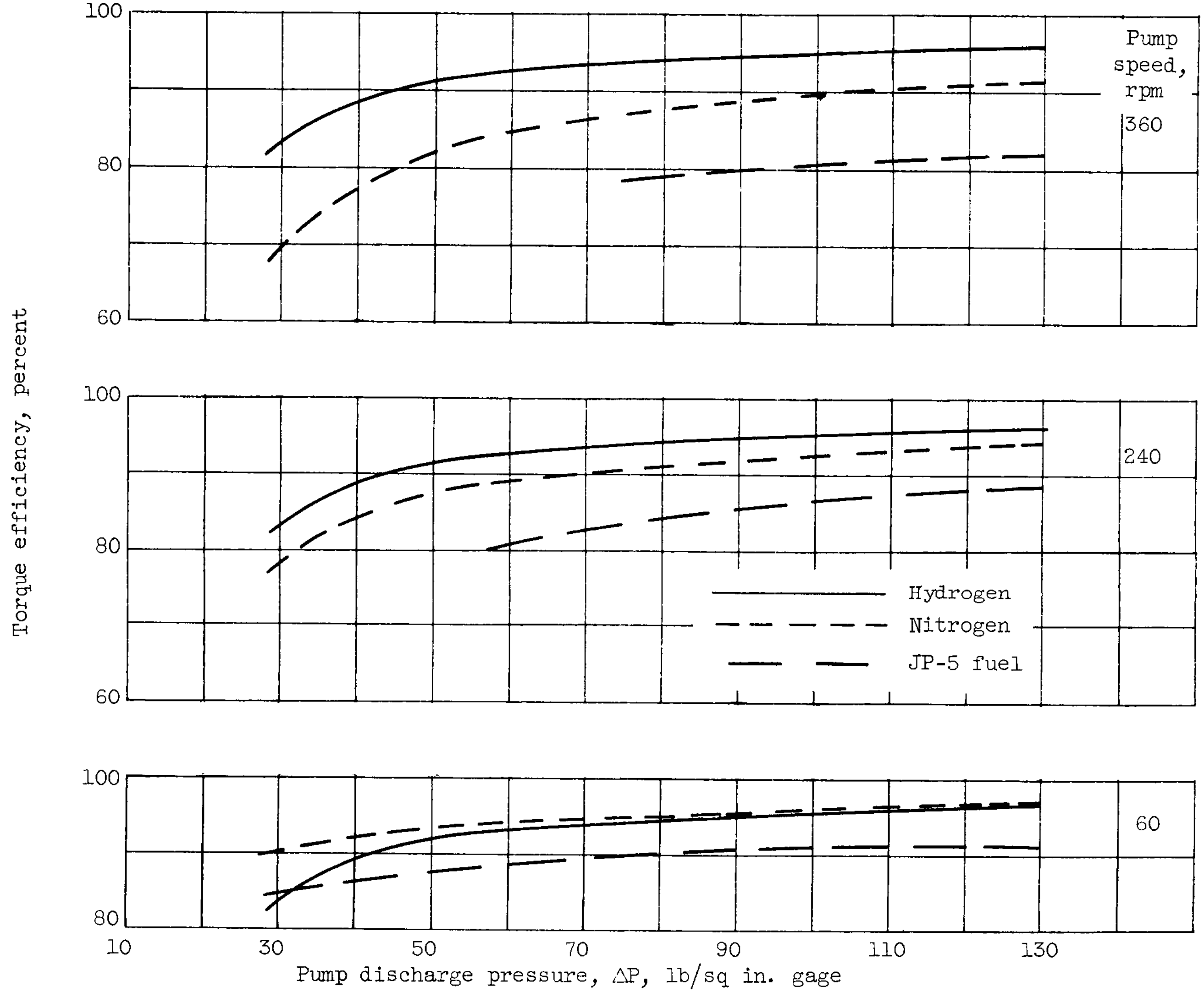


Figure 13. - Torque efficiency with JP-5 fuel, boiling hydrogen, and boiling nitrogen.

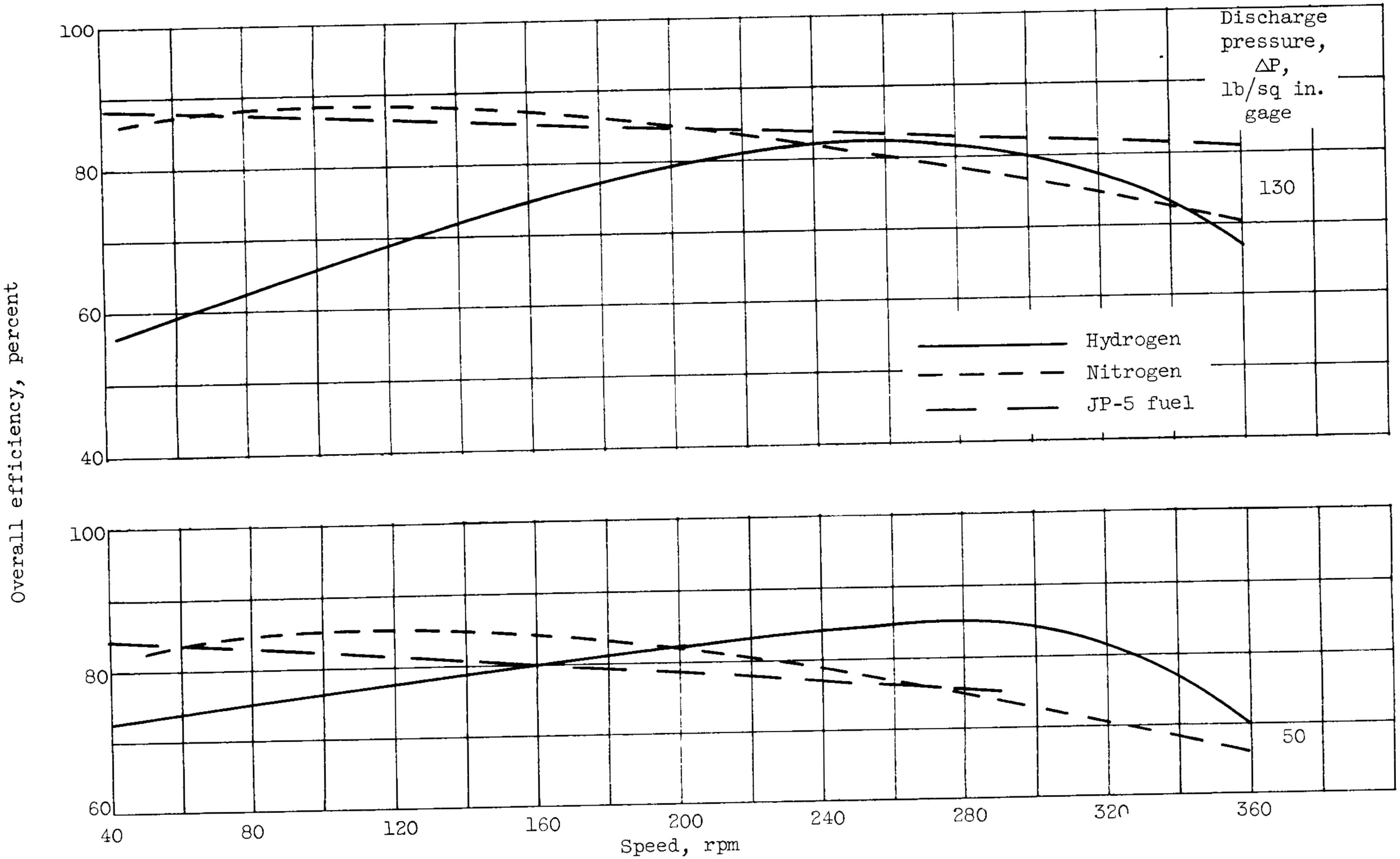


Figure 14. - Overall efficiency with JP-5 fuel, boiling hydrogen, and boiling nitrogen.